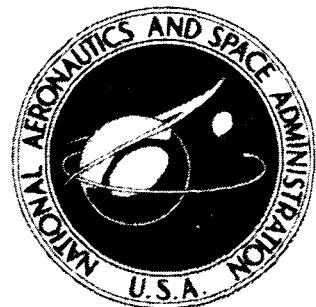


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ROTATING PRESSURE MEASURING SYSTEM
FOR TURBINE COOLING INVESTIGATIONS

by Frank G. Pollack, Curt H. Liebert,
and Victor S. Peterson

Lewis Research Center
Cleveland, Ohio 44135

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • OCTOBER 1972

1. Report No. NASA TM X-2621	2. Government Accession No.	3. Recipient's Catalog No.	
4. Title and Subtitle ROTATING PRESSURE MEASURING SYSTEM FOR TURBINE COOLING INVESTIGATIONS		5. Report Date October 1972	
		6. Performing Organization Code	
7. Author(s) Frank G. Pollack, Curt H. Liebert, and Victor S. Peterson		8. Performing Organization Report No. E-7016	
		10. Work Unit No. 501-24	
9. Performing Organization Name and Address Lewis Research Center National Aeronautics and Space Administration Cleveland, Ohio 44135		11. Contract or Grant No.	
12. Sponsoring Agency Name and Address National Aeronautics and Space Administration Washington, D. C. 20546		13. Type of Report and Period Covered Technical Memorandum	
15. Supplementary Notes		14. Sponsoring Agency Code	
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17. Key Words (Suggested by Author(s)) Rotating pressure measurement Turbine cooling Rotating machinery Compressors		18. Distribution Statement Unclassified - unlimited	
19. Security Classif. (of this report) Unclassified	20. Security Classif. (of this page) Unclassified	21. No. of Pages 19	22. Price* \$3.00

* For sale by the National Technical Information Service, Springfield, Virginia 22151

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SUMMARY

The development of a 10-channel rotating pressure measuring system capable of operation to speeds of 9000 rpm at transducer temperatures of about 320 K (120° F) is described. Variable-reluctance pressure transducers were mounted in the rotating system for sensing pressure.

Rotating performance tests on a spin rig showed that the output data from 7 of the 10 transducers tested were within a desired system error of 3 percent. However, the error of the output data from three other transducers was as large as 8 percent at 9000 rpm. It was concluded from these test results that a rotating screening method was necessary to evaluate each pressure transducer channel within a system that will be used under rotating conditions.

INTRODUCTION

This report describes the development of a 10-channel rotating pressure measuring system capable of operation to shaft speeds of 9000 rpm and environmental temperatures of about 320 K (120° F). Such systems have long been desired for making pressure measurements on rotating components. Two applications for rotating pressure systems are the measurement of the static-pressure distribution on the surfaces of rotating turbine blades and the measurement of the total and static pressures of a coolant flowing within rotating turbine blades. The former application could provide data for more accurate turbine design calculations and better estimates of external turbine blade heat-transfer coefficients. The latter application could be used to measure the coolant flow within air-cooled turbine blades.

Early attempts to measure pressures on the surfaces of rotating compressor and turbine blades have made use of cumbersome mechanical devices (refs. 1 and 2). More

recent efforts in the field of rotating pressure measurements (ref. 3) have used a combination of electromechanical transducers and electronic instrumentation and have resulted in a greater accuracy with less size than the totally mechanical devices. The general requirements for a pressure transducer operating in a rotating environment are that its performance be relatively insensitive to environmental stimuli such as centrifugal forces or temperature fluctuations. The variable-reluctance differential-pressure transducer described in reference 4 is a good candidate for fulfilling these requirements. This transducer was incorporated in the design of the current system.

The rotating pressure measuring system described in this report was designed for obtaining steady-state pressure data from air-cooled turbine blades operating in an experimental turbojet engine at the Lewis Research Center. The rotating performance of this pressure measuring system was investigated in a spin test rig at the environmental speed and temperature that would be encountered by the system when operating with this engine. Pressure data obtained from the rotating system are compared with data from a nonrotating reference pressure probe. The design goal was a measurement system error of 3 percent or less at 9000 rpm.

ROTATING PRESSURE MEASURING SYSTEM

A 10-channel rotating pressure measuring system consisting of a rotary package and stationary instrumentation was designed for obtaining pressure data from air-cooled turbine blades operating in an experimental turbojet engine. The rotary package could operate at shaft speeds up to 9000 rpm. Figure 1 illustrates schematically the location of the rotary package on the engine, the stationary instrumentation, and the path of the pressure tubing extending from the pressure taps on the turbine to the rotary package.

A photograph of the rotary package is shown in figure 2. The package was about 37 centimeters (15 in.) long and 17 centimeters (7 in.) in diameter. It consisted of a stationary outer housing, a rotating bulkhead assembly containing the differential-pressure transducer, calibration ports, pressure tubing inlet ports (not shown), and a sliring assembly. The transducers were used to convert a pneumatic pressure force into an electrical signal. The sliring assembly transferred excitation voltage from a stationary power supply to the rotating transducers. It also transferred output signals from the transducers to stationary carrier amplifiers. The design goal was a measured system error of 3 percent or less under maximum speed conditions of 9000 rpm. Details of the rotary package design are discussed in the next section.

ROTARY PACKAGE

Variable-Reluctance Transducers

The transducers selected for the rotary pressure package were the variable-reluctance pressure difference type described in detail in reference 4. This type of transducer was selected for this application after a variety of commercially available pressure transducers were tested. This transducer was available in a variety of differential pressure ranges from 0 to 1.4 newtons per square centimeter (0 to 2 psi) to 0 to 70 newtons per square centimeter (0 to 100 psi). The overall size of the transducers was the same for all pressure difference ranges. The transducers were approximately 1 centimeter (7/16 in.) in diameter and 0.6 centimeter (1/4 in.) thick. As discussed in reference 4 these variable-reluctance transducers are capable of measuring pressure differences when operating at centrifugal accelerations up to 5000 g's and at temperatures of 230 to 370 K (-50° to 200° F).

The construction of the variable-reluctance, pressure difference transducer is illustrated in simplified form in figure 3. The transducer consisted of two sturdy symmetrical case halves separated by a diaphragm. The case and diaphragm were made from a nickel-iron magnetic alloy. Each case half was a circular cup core in cross section and contained an inlet pressure port on its centerline. An insulated inductance coil made of manganin wire was secured in each case half. An air gap existed between the cup core and the diaphragm. External to the case, the electrical leads from the coils were encased in a braided shield and potted to form a semirigid cable.

Figure 4 is a block diagram of the 10-channel rotating pressure measuring system. Each transducer was excited by the carrier oscillator power supply of 5 volts at 20 kilohertz. Within each transducer the two coils were connected in series and formed two arms of a four-arm bridge circuit. The two other arms of the bridge circuit were internal to a stationary carrier amplifier. When a pressure difference was applied to the transducer, the diaphragm deflected and thereby changed the magnetic reluctance coupling between the two case halves. This unbalance changed the inductance of the coils which was sensed by the bridge circuit. The signal was transferred electrically through slip-rings to the stationary carrier amplifier and the data recording system.

Each transducer was inspected and tested for mechanical integrity and electrical characteristics, which included coil resistance and resistance to ground. In addition, pressure calibrations at nonrotating conditions were performed on each transducer at various transducer temperatures in the range of interest. A precision gage calibrated to 0.25 percent was used to indicate pressure. The electronic measuring system was accurate to within 0.5 percent. As a part of the calibration procedure, zero shift, hysteresis, and sensitivity shift were determined. Zero shift is defined as the change with

transducer temperature in output voltage with zero pressure difference applied to the transducers. Hysteresis is the maximum difference between the output voltage readings at any pressure level where the level is approached with increasing and then decreasing pressure during any one calibration cycle. Sensitivity shift is the change in slope with transducer temperature of a straight line connecting the average zero pressure point and the full-scale pressure point.

Slipring Assembly

The commercially available slipring assembly used in this study provided a mercury-wetted interface between each rotating ring and its associated stator ring. A pool of mercury within each ring compartment was centrifuged during rotation. This centrifuging produced a continuous contact around the periphery resulting in a noiseless electrical contact between the slipring rotor and stator. The slipring assembly contained 13 signal transfer rings. Two rings carried the common carrier oscillator power supply to the 10 transducers. Ten rings were used to transfer the individual outputs of the transducers to individual stationary carrier amplifiers. The remaining ring was used as a common system ground link.

Package Mechanical Design

The bulkhead assembly shown in figure 2 was made of aluminum and consisted of transducer compartments, a mounting plate, and a pressure tubing compartment. Each transducer was mounted in a separate transducer compartment, so that the transducer diaphragm was in the plane of rotation. The transducer centers were separated from each other by 36° and were mounted 2.86 centimeters (1.12 in.) off the shaft centerline. The transducer compartments were easily removable from the bulkhead assembly for convenient installation or replacement of the transducers. Figure 5 shows one of the transducers removed from its compartment. Both the compartment and the transducer are turned 90° with respect to the face of the mounting plate for clarity of observation. The transducers are shown wired to the slipring assembly. The cooling fan (fig. 5) dissipates heat away from the transducers. This heat is generated by the slipring during rotational operation of the package.

The transducer compartments were attached to one side of the mounting plate, also shown in figure 5. Each transducer compartment was secured to the mounting plate with four stainless-steel screws so that no movement could occur during high-speed rotation. The interfaces between the transducers and the mounting plate and compartments were

sealed with O-rings. The clearances were adjusted so that the O-rings were compressed properly to seal the pressure, yet no strain was placed on the transducer case. Shims were used to compensate for slight differences in case thickness. A strain on the transducer case can cause either a zero shift or a sensitivity change or both. Extreme care was used during transducer installation to eliminate case strain. Pressure calibrations were rechecked after the transducers were installed in the transducer compartments to verify proper installation.

Mated to the other side of the mounting plate with five bolts was the pressure tubing compartment. Figure 6 shows this compartment separated from the mounting plate. This separation was required when the package was first installed and is discussed again in the section Spin Rig. This compartment contained 10 pairs of inlet ports, 1 pair for each transducer. Pressure tubes extending from pressure measuring points terminated at these ports. The inlet ports had tapped holes for 0.32-centimeter (1/8-in.) compression fittings which could accommodate either 0.10- or 0.15-centimeter- (0.04- or 0.06-in.-) diameter pressure tubing.

Figure 7 is a cutaway drawing of the assembled rotary package. It shows the location of the transducers as well as the transducer compartments, the mounting plate, and the pressure tubing compartment, which compose the bulkhead assembly shown in figure 2. The pressure tubing inlet ports were connected to the transducers by means of a series of mating drilled passages through the bulkhead assembly as shown in the figure. Pilot tubes were inserted in the drilled passages on the mounting plate; around each pilot tube was an O-ring to seal the pressure transfer junction. These pilot tubes and O-rings can also be seen in figure 6. The mounting plate had a stub shaft (fig. 7) that was used to couple it to the sliring assembly shaft. Before assembly, all parts and passages were ultrasonically cleaned. After assembly the rotating parts were leak checked and balanced.

Rotary Package Assembly

The rotary package shown in cross section in figure 7 was assembled in the following sequence. The front-end parts, the shaft coupling, bearing assembly, and pressure tubing compartment (on the right in fig. 7), were first mated in a subassembly. This subassembly was then mounted to the stationary engine case or test rig on which the package was to be operated. Pressure tubing from the pressure taps on the engine or rig was passed through the hollow shaft and attached to the proper inlet port on the pressure tubing compartment (see figs. 6 and 7). Then the subassembly (consisting of the mounting plate, transducer compartment, and sliring assembly) shown in figure 5 was bolted to the pressure tubing compartment. Next, the larger part of the stationary outer housing with the bearing assembly shown on the left in figure 7 was installed and bolted

to the smaller part of the outer housing. As this mating of parts was made, the shaft of the sliring assembly fit into a keyed bearing adapter to complete the continuity of shafting through the rotary package. The stator housing of the sliring assembly was clamped to the outer housing of the rotary package to prevent rotation.

In addition to the details just described, several other important features of the rotary package are worth noting. The rotary package was designed to mount externally on an engine and rotate with the engine shaft. The bulkhead assembly (mounting plate, transducer compartments, and pressure tubing compartment) was massive to ensure rigidity and temperature uniformity for the transducers and to prevent rapid changes in transducer temperature. A hollow shaft throughout the package allowed thermocouple leads to extend from the engine, through the shaft of the rotary package, and to a rotating thermocouple sliring device (or a shaft data system such as described in ref. 5) mounted in tandem with the package. The shaft data system would measure hot turbine blade and disk temperatures and transducer temperatures. Finally, each transducer could be calibrated at nonrotating conditions after the package was assembled by using the calibration ports provided in the pressure tubing compartment (fig. 7). These ports can also be seen in figure 2.

SPIN TESTING

The rotating performance of the entire rotating pressure measuring system was investigated at the environmental speed and temperature that would be encountered by the system when operating with the research turbojet engine at the Lewis Research Center. The investigation was conducted by mounting the rotary package on a spin rig which carried a suitable array of pressure sensing taps. The measurement accuracy of each transducer channel was evaluated with a simple screening method. This method is discussed in the section Screening Test Procedure. Transducers with pressure ranges of 0 to 34, 0 to 17, and 0 to 6.9 newtons per square centimeter (0 to 50, 0 to 25, and 0 to 10 psi) were used. Before the spin tests, and at room temperature, the sensitivity of each transducer channel amplifier was adjusted so that a pressure range of 0 to 3.4 newtons per square centimeter (0 to 5 psi) gave an output range of 0 to 10 volts dc. Then precision calibrations between 0 and 4.1 newtons per square centimeter (0 to 6 psi) were made for each transducer channel at the elevated test temperatures of 312 and 328 K (102° and 130° F). Repeated calibration cycles were within 1 percent of full-scale output for the 0- to 4.1 newton-per-square-centimeter (0- to 6-psi) calibration range.

Spin Rig

A cold-air, turbine-driven spin rig encased in the protective structure shown in figure 8 was used to spin the rotary package at speeds up to 9000 rpm. The wheel on this spin rig was 61.5 centimeters (24.2 in.) in diameter and had a hollow shaft adapter which was bolted to the hub on the front face. The outer housing of the rotary package was bolted to the spin rig. The package shaft was coupled to the spin rig shaft. Secured in the rim of the wheel was a test target with its outer face flush with the curvature of the wheel. On the face of the target were 10 pressure taps drilled radially downward through the test target. A pressure tube 0.1 centimeter (0.04 in.) in diameter was brazed into the base of the target in line with each of the 10 pressure taps. These tubes extended radially inward along the face of the wheel (to which they were rigidly secured with several tack welded strips) through the hollow shaft adapter and were connected to the inlet ports in the pressure tubing compartment as shown in figure 8. (The remainder of the rotary package had not yet been assembled when the photograph in fig. 8 was taken.) Each tube was connected to a port on one of the transducers by means of the drilled passages in the pressure tubing compartment. The other port on each transducer was connected to a common tube which extended through the hollow shaft of the rotary package. This common tube sensed the barometric pressure at the centerline of the package outside the enclosed spin rig. After the tubing connections were completed, the remainder of the rotary package was assembled as described in the section Rotary Package Assembly. A stationary pressure probe (fig. 8) was installed close to the wheel rim and was connected to a water manometer. The pressure measured by this probe was used as the reference pressure (p_{ref}) during the rotating tests. This reference probe was located at the target radius at a spacing of approximately 0.5 centimeter (0.2 in.) from the side of the wheel.

For the spin test, the rotary package slipring assembly was connected to the stationary part of the system (fig. 4) with a 46-meter (150-ft) cable to simulate the actual length necessary for future test cell engine testing. This cable consisted of 10 single-conductor shielded wires and one two-conductor twisted shielded pair. All conductors were 16 gage wire. The amplifiers were connected to a data acquisition system consisting of an integrating digital voltmeter, a cross bar scanner, and a printer. In addition, 10 voltmeters were used to observe continuously the voltage outputs from the 10 individual transducers. A strip-chart recorder was used to obtain a permanent continuous record of a selected transducer output.

Screening Test Procedure

The arrangement of the transducers in the spin apparatus is shown schematically in

figure 9. The transducers were located at a radius of 2.86 centimeters from the centerline. One side of the transducer diaphragm was exposed to atmospheric pressure p_{bar} in the test cell. The other side was exposed to the sensed pressure at the target p_t . The sensing taps on the target were at a radius of 30.73 centimeters from the centerline. When the apparatus was not rotating, p_{bar} was equal to p_t . At rotating conditions this relation was still essentially true. Because of centrifugal forces on the air columns, rotation of the system produced increasing values of the pressure difference across the transducers $p_{\text{II}} - p_{\text{I}}$ with increasing shaft speeds. In figure 9, p_{II} and p_{I} are the pressures on each side of the transducers. During rotational testing, transducer voltage outputs of the transducers were recorded, and the values of the quantity $p_{\text{II}} - p_{\text{I}}$ were determined from corresponding calibration curves from nonrotating tests. Then p_t was calculated from these values and compared with the pressure measured by the stationary reference probe p_{ref} . In this way, the accuracy of each transducer channel under rotating conditions was evaluated.

Calculations for the pressure at the target involved use of the following general relation (ref. 1) to account for the centrifugal force (pressure) on the air columns in the tubes:

$$p_A = p_B \exp \left\{ \frac{\omega^2 [(r_A)^2 - (r_B)^2]}{2gRT} \right\} \quad (1)$$

where

p_A pressure at r_A

p_B pressure at r_B

ω angular velocity, $\frac{2\pi N}{60}$

N shaft speed, rpm

r_A distance of p_A from centerline

r_B distance of p_B from centerline

g acceleration due to gravity

R gas constant of air

T temperature of air in pressure tubes connecting p_A and p_B

This equation assumed a one-dimensional balance of forces on the air column, the perfect gas law, and no temperature gradient along the air column. The calculation procedure using the International System of Units was as follows. First, p_{II} (fig. 9) was calculated by substituting $r_B = 0$, $r_A = 2.86$ centimeters (1.12 in.), $p_B = p_{bar}$, $T = T_{II}$, and $p_A = p_{II}$ into equation (1):

$$p_{II} = p_{bar} \exp \left[1.560 \times 10^{-8} \left(\frac{N^2}{T_{II}} \right) \right] \quad (2)$$

The temperature T_{II} is the temperature (K) of the air column connecting p_{II} with p_{bar} , and p_{bar} is the barometric pressure. The pressure p_I was then calculated from the experimental values of the quantity $p_{II} - p_I$. Then p_t was calculated by substituting $r_B = 2.86$ centimeters (1.12 in.), $r_A = 30.73$ centimeters (12.1 in.), $p_B = p_I$, $T = T_I$, and $p_A = p_t$ into equation (1):

$$p_t = p_I \exp \left[1.787 \times 10^{-6} \left(\frac{N^2}{T_I} \right) \right] \quad (3)$$

where T_I is the temperature (K) of the air column connecting p_T with p_I .

The temperatures of the air columns T_I and T_{II} could not be directly measured because a rotating thermocouple pickup was not available. The wheel and the pressure tubes attached to it were enclosed separately from the rotary package. The wheel enclosure air temperature and package enclosure air temperatures were measured with thermocouples, and data were recorded several minutes after steady-state enclosure temperatures were reached at selected shaft speeds. Calibrations taken prior to testing showed that temperatures and speeds are known within 0.1 percent. The tubes were thin-walled and lay on the surface of the wheel, where they were in direct contact with the air in the wheel enclosure. The temperature T_I of the air columns in the 10 pressure tubes mounted on the wheel were assumed to be within 3 K (5° F) of the enclosure air temperature. Also, the temperature T_{II} was assumed to be within 3 K (5° F) of the rotary package enclosure temperature. An uncertainty of 3 K will result in a maximum error of 0.5 percent in the calculated target pressure.

The air in the separate enclosure surrounding the rotary package was heated with an electric heater. The air temperature in this enclosure was maintained at preset temperature levels of 312 and 328 K (102° and 130° F) during two separate tests. Before the static (nonrotating) calibration runs and the spin tests, these temperature levels were maintained for at least 2 hours to ensure that the pressure transducers reached the

same temperature as the surrounding air. Thermocouples were mounted on the transducer compartments to monitor the temperature during nonrotating conditions. A static calibration at each of the two spin test temperatures was made directly before and immediately after the spin tests.

RESULTS AND DISCUSSION OF SPIN TESTS

The performance of the pressure measuring system under rotating conditions was evaluated by comparing the 10 pressures sensed dynamically at the target on the wheel rim with the pressure directly sensed by using the stationary reference probe mounted very close to the wheel surface. The reference probe pressures are known within 0.1 percent.

Figures 10 and 11 present the system error (percent deviation of p_t from the reference pressure) as a function of shaft speed at two transducer environmental temperatures. At a transducer temperature of 312 K (102° F) data are presented for shaft speeds between 6000 and 9000 rpm (fig. 10). At a transducer temperature of 328 K (130° F) data are presented for shaft speeds between 3000 and 9000 rpm (fig. 11). The data were taken at steady-state speed conditions at speed increments of about 1000 rpm on both increasing and decreasing speed sequences.

The pressure measurement system design goal was a 3-percent deviation of p_t from the reference pressure at the maximum shaft speed of 9000 rpm, and most of the data fell within this percentage, (figs. 10 and 11). However, some target pressures deviated from the reference pressure by as much as 8 percent. Error was least at the lower speeds and generally increased as shaft speed was increased. For many turbine cooling applications, an error within 5 percent is acceptable.

The reasons for the deterioration of rotational performance of the system at the higher shaft speeds were not investigated in detail in this study, but the data scatter was probably the result of centrifugal force effects on various transducer components. With the centerline of the transducers on a 2.86-centimeter (1.12-in.) radius, a force of 2600 g's was experienced on the diaphragm at 9000 rpm. Adverse effects of centrifugal forces on transducer diaphragms have been documented in the literature (ref. 3). For this small sampling, there is little correlation between transducer ranges and system error. When the transducer channels were calibrated under nonrotating conditions, the calibration cycle repeatability was consistent and was within 1 percent of full-scale output. These nonrotating calibration tests did not reveal the wide variation of output (8 percent) that occurred in the rotating tests. This was a typical observation of many other spin tests and appears to be unpredictable from any nonrotating tests or inspections made on the transducer and from the method of containing the transducer within the

system. Therefore, it appeared that a rotating screening method similar to the one described in this report is necessary to evaluate pressure transducer channels that are to be used under rotating conditions.

CONCLUDING REMARKS

The performance of a 10-channel rotating pressure measuring system was investigated at temperatures of about 320 K (120° F). Under nonrotating conditions, the system error of each channel, as well as the calibration cycle repeatability, was within 1 percent of full-scale output. However, at maximum shaft speeds of 9000 rpm test results showed that the measure of system error of the individual channels varied from 3 to 8 percent. The output data of seven transducer channels were within the desired system error of 3 percent. However, the error of the output data from three other transducer channels was as large as 8 percent. It was concluded that nonrotating tests were not sufficient verification of the performance of the system under rotating conditions. A rotating screening method is necessary to evaluate pressure transducer channels that will be used in a system under rotating conditions.

Lewis Research Center,
National Aeronautics and Space Administration,
Cleveland, Ohio, June 30, 1972,
501-24.

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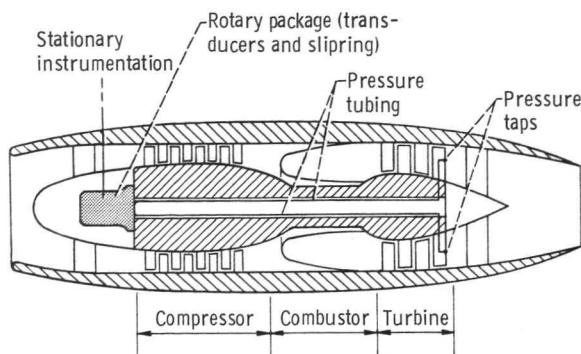


Figure 1. - Schematic of engine - rotating - pressure - measuring - system configuration.

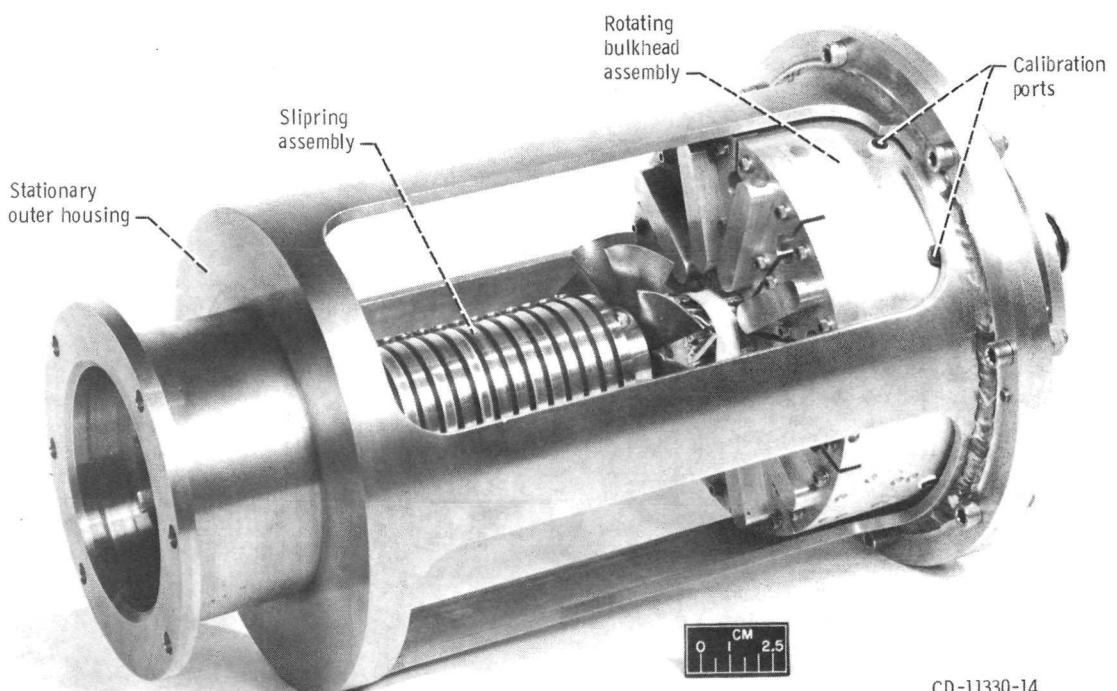


Figure 2. - Rotary package.

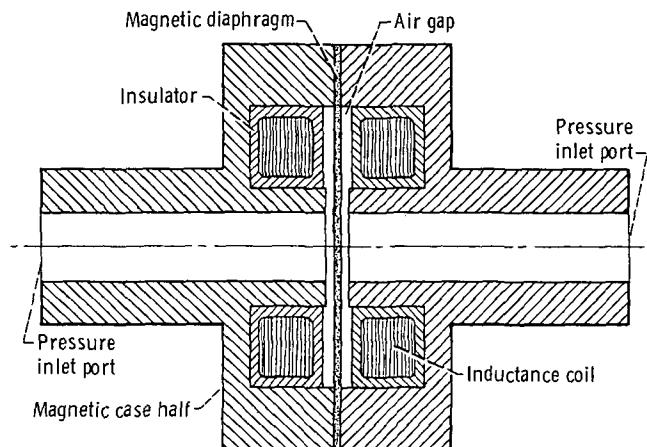


Figure 3. - Variable-reluctance differential-pressure transducer.

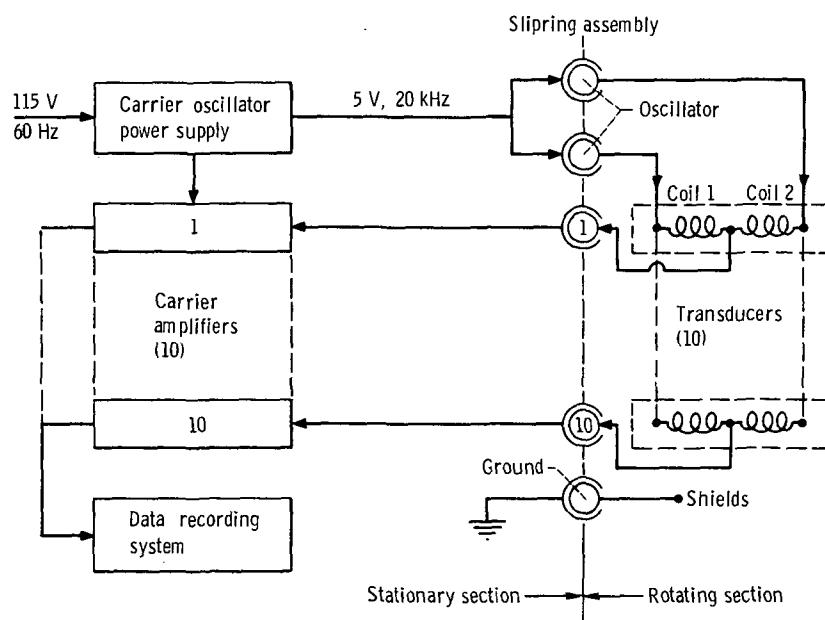


Figure 4. - Block diagram of 10-channel rotating pressure measuring system.

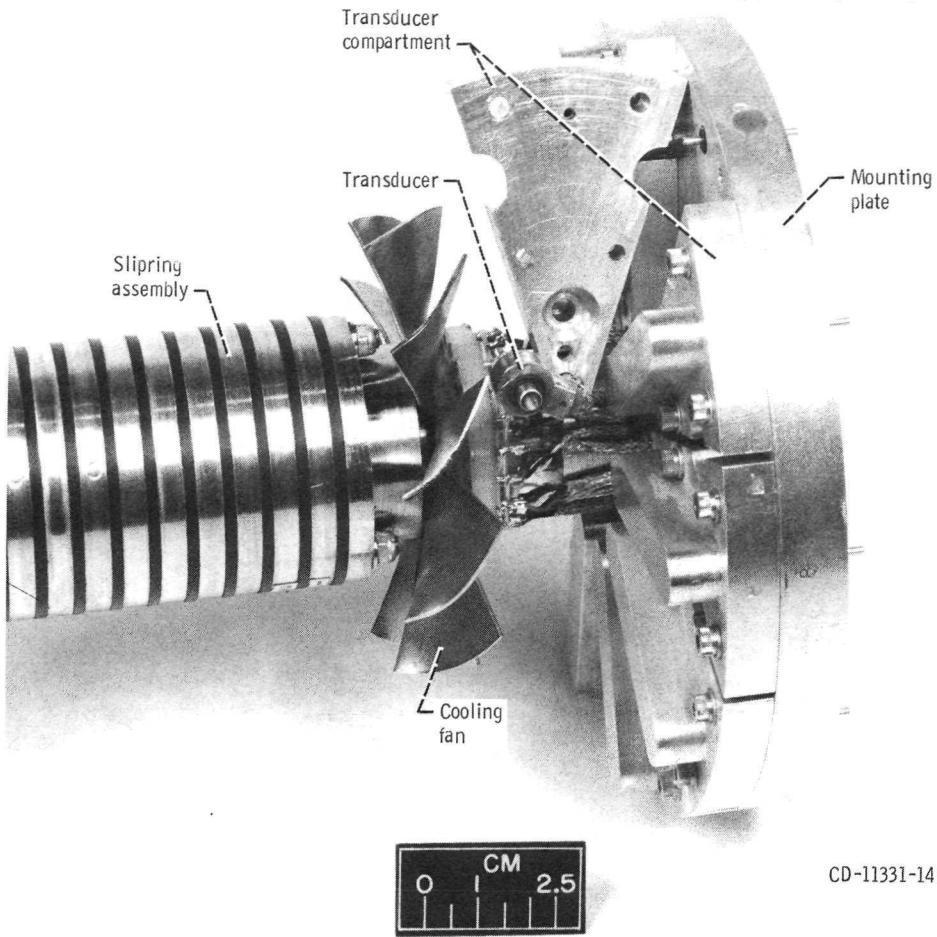


Figure 5. - Transducer and transducer compartment attached to mounting plate.

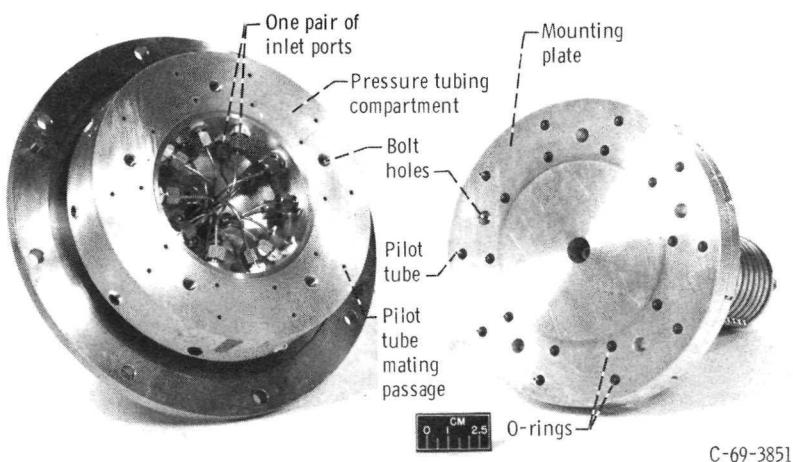


Figure 6. - Pressure tubing compartment and mounting plate.

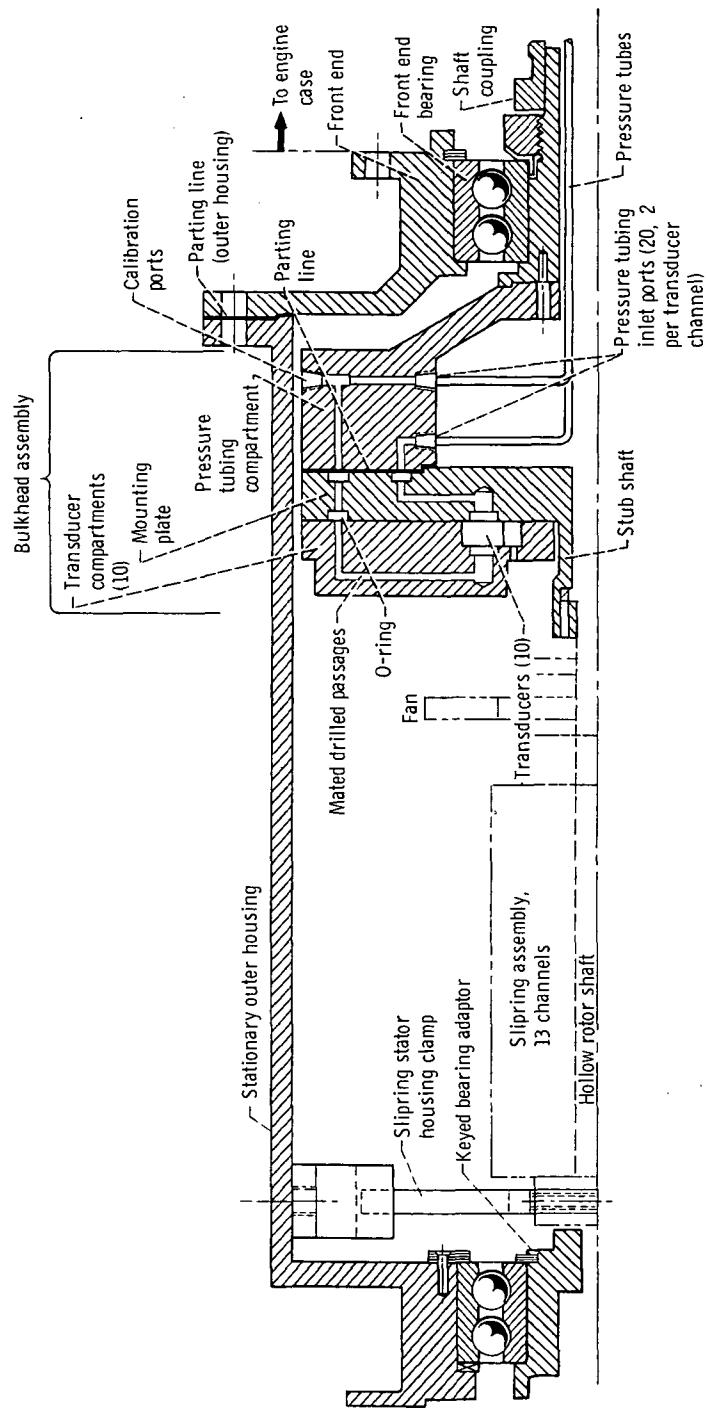


Figure 7. - Cutaway drawing of assembled rotary package.

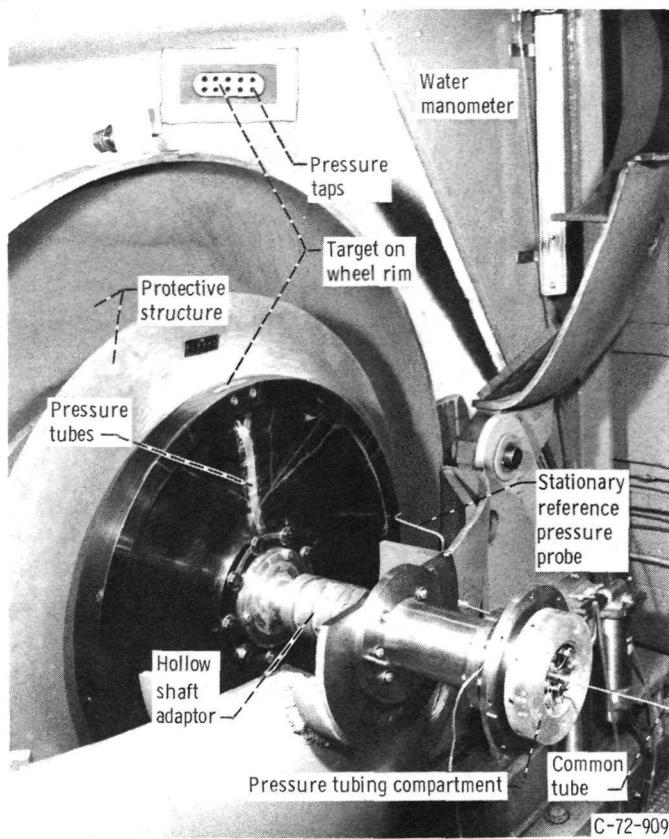


Figure 8. - Spin rig and test target.

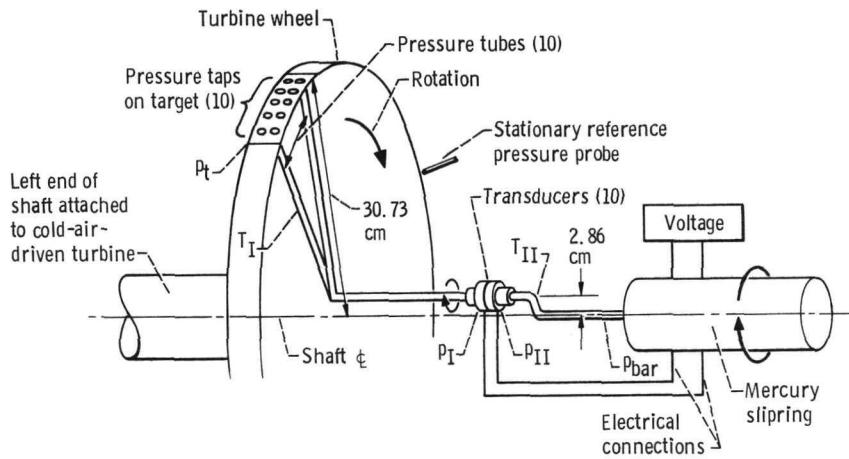


Figure 9. - Schematic of spin apparatus.

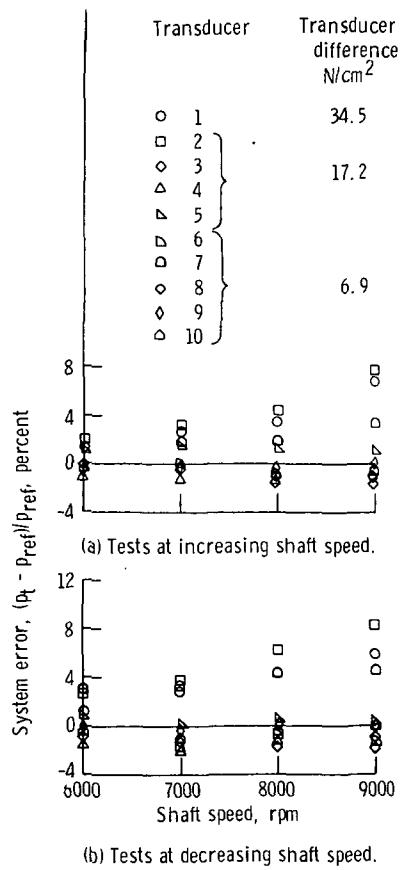


Figure 10. - Rotary-pressure-measuring-system performance at 312 K (102° F).

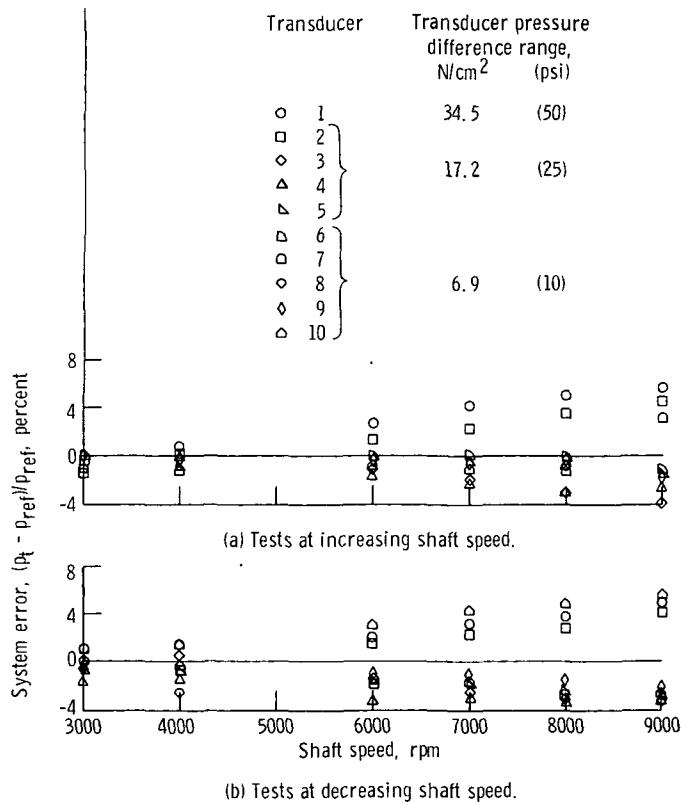


Figure 11. - Rotary-pressure-measuring-system performance at 328 K (130° F).

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